FASTENERS

VITAL IN EVERY INDUSTRY

PUBLISHED BY AMERICAN INSTITUTE OF BOLT, NUT, AND RIVET MANUFACTURERS

VOLUME ONE • NUMBER ONE

170 million pieces DAILY is the average output of the Fasteners Industry. Made to close tolerances, they are uniform and dependable.

HOW THE AMERICAN INSTITUTE OF *Bolt, Nut and Rivet Manufacturers* *can help USERS of fasteners*

For many years, the AMERICAN INSTITUTE OF BOLT, NUT AND RIVET MANUFACTURERS has functioned as a clearing house for technical problems in connection with the application and use of headed and threaded products. The Institute has cooperated with other organizations in developing standards of dimensions of the products of the industry to the end of complete interchangeability. A handbook of Bolt, Nut and Rivet Standards has been widely distributed to users as a basic aid to design and assembly. It advises its membership in regard to laws, regulations and rulings, and is continually in active contact with government bureaus and the armed services.

To broaden its service to users of fasteners, the Institute is inaugurating a program of technical research. A number of subjects never before thoroughly investigated have been selected and the first project is already under way at the Technological Institute of Northwestern University. The facilities available for this work are illustrated on pages 13 to 15.

The Institute is represented on many engineering and business committees and maintains a capable headquarters engineering staff to serve members and users of fasteners. It includes in its membership the leading and most active manufacturers of bolts, nuts, rivets, screws and other headed and threaded products.

Problems brought to the Institute by users of fasteners are either answered by the Technical Staff or referred to manufacturers whose engineers have had the necessary experience.

We sincerely hope you will like our initial effort as expressed in this first issue of "FASTENERS", and that you will retain this and future issues in your files for ready reference. Our primary objective, of course, will be to provide in this publication the factual engineering data you and other executives need to keep abreast of the important developments in this basic industry.

Your comments and suggestions will be welcomed.

AMERICAN INSTITUTE OF
BOLT, NUT AND RIVET MANUFACTURERS

Herman N. Reid
President

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The Rolled Screw Thread Process

Opens Bottlenecks

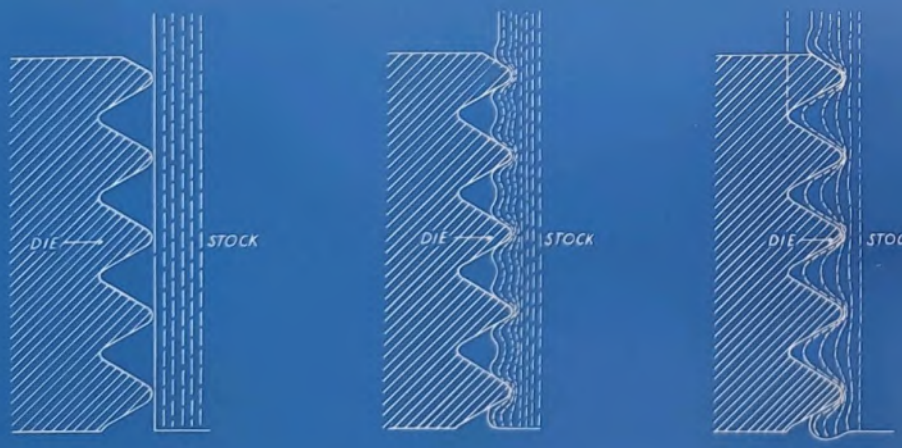
by George S. Case

Chairman of the Board, The Lamson and Sessions Co.

. . . . continued improvement in the process of roll threading, resulting in greater accuracy, had led to its use in precision work . . . . replacing ground threads on aircraft bolts and studs, the process eliminated a potential bottleneck in aircraft production.

BECAUSE of the war demand for economical production of large quantities of bolts, screws, studs, and other threaded fasteners, screw threads produced by roll threading have recently been of great interest. Roll threading is usually a cold forming process whereby a bolt or other cylindrical object has threads formed on its external surface by means of squeezing between rotating or reciprocating dies. The action of the dies on the metal causes the surface to be indented by the crest of the die threads while the metal between the die threads is forced outward to form the apex of the thread.

Although roll threading has come into extensive use only in the last decade or two, the roll threading process was first used even before bolts were manufactured commercially, the earliest recorded use of rolled threads dating back to 1851, with some evidence to indicate use of this method even earlier. The earliest rolled threads were made by holding a grooved roll against a bar held in a lathe. Later, machines were developed to roll threads on bolts and screws. Some thread rollers were made in the 80's with two flat dies, both moving, but in opposite directions. Another type of thread roller used two circular dies, one inside a stationary ring and the other on the edge of a rotating disc. Still other machines were developed using two disc shaped dies, rotating in opposite directions so that the bolt being threaded rotates in the same position at the point where the dies are closest together. The



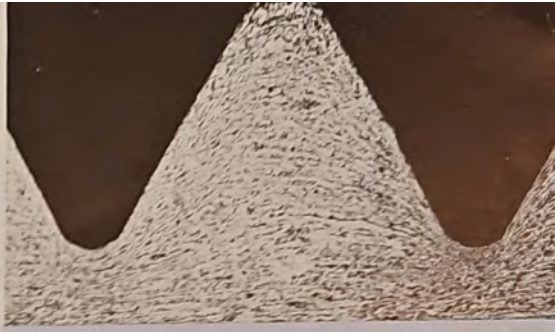
Three steps in the action of the die in producing rolled threads. The lines extending through the bolt stock are indicative of the directional lines of flow in the steel and show how the process affects them.

machine now in almost universal use for tonnage production is a horizontal type roller with one flat die held stationary and the other reciprocated across it.

Much of the progress and improvement of the roll threading process must be credited to the development of better grades of material. When roll threading was first introduced to manufacturers of bolts and screws, wrought iron was used. Thread rolling was extremely difficult as the iron had a tendency to split under the pressure of the dies. The introduction of Bessemer steel wire to the bolt industry helped to some extent as bolts that could be readily headed of that comparatively hard material could also be roll threaded. Since the development of open hearth steel, which can be cold headed and cold rolled into almost any type of bolt threads, rolling has become almost universal practice for large production of small and medium sized bolts and screws, although it is still not economical to roll thread very large diameters or very long bolts. It is estimated that approximately 95 per cent of the total number of bolts and screws now being made have rolled threads.

The introduction and development of roll threading as in any other method of manufacture was undoubtedly helped by the fact that it was an economical process for high production. The production rates possible with roll threading are from 5 to 100 times faster than cutting or grinding. Considerable saving in material is possible because of the fact that no material is removed to be lost as scrap.

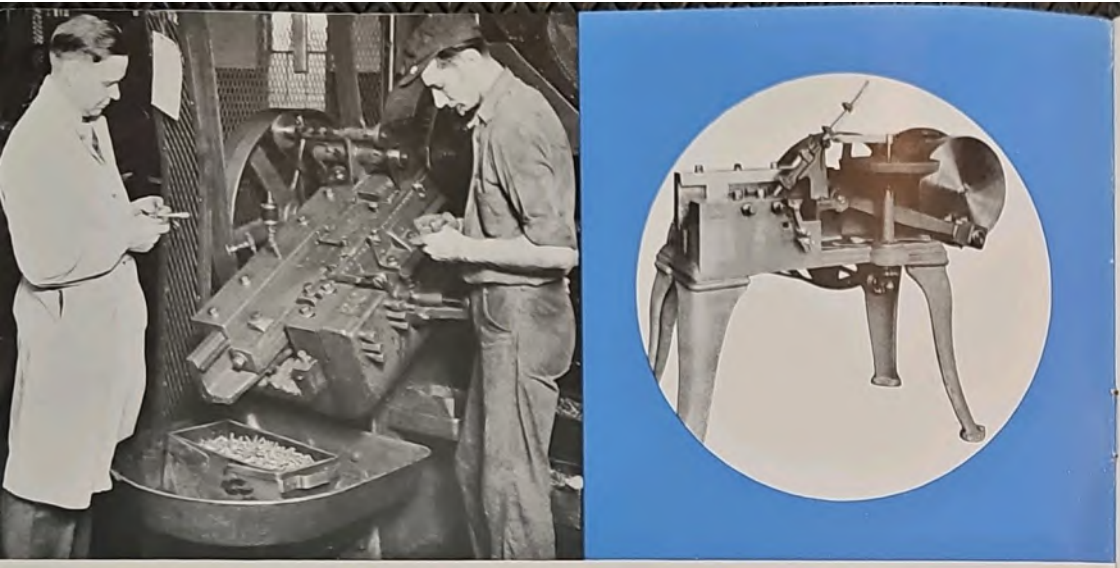
Stove bolts, machine screws and tire bolts were made with rolled threads from 1880 on but it was not until 1900 that bolt manufacturers brought out rolled thread carriage bolts as a second low priced competitive line. These bolts were made from wire of approximately the pitch diameter of the screw threads so that the outer diameter of the threads after rolling was larger than the shank, resulting in a bolt with an undersized shank (see sketches). Since in most cases carriage bolts are used in wood and the bolts have a square under the head, the undersized shank was not objectionable. Later on, lines of rolled thread machine bolts and other common bolts were brought out. Since these were put out as the cheapest possible bolt they were frequently not well fitted nor well finished from an appearance standpoint and it was not customary to apply the nuts to the bolts when they were sold in packages. As a result the rolled thread label became a mark of an inferior product and jobbers and bolt users making high grade products refused to have anything to do with them.



Photomicrograph showing structure of a cold rolled thread.

In a relatively short time, however, it was found that rolled threads could be made more accurately than cut threads with the same degree of care in manufacturing. It was also found in products made of soft steel and some non-ferrous metals that the cold working of the metal improved the quality. At present bolts bearing the rolled thread label are as well made as any other bolts. It is hard to think of any application where rolled thread carriage bolts will not do the job required and rolled thread machine bolts are economical and satisfactory in most places.

Improvements in cold heading methods and materials made it possible to upset the shank of the shorter bolts between the threads and the head so that the threads could be rolled and the shank remain full size. When this bolt was pointed its appearance was so nearly the same as a bolt having a machine-cut thread that there was great difficulty in distinguishing between them. This bolt, sold under a cut thread label, soon became popular and is the bolt most used today.



Modern mass production machines turn out roll threaded bolts, studs and screws with a precision undreamed of when the early machines (in circle) were in vogue.

A still later development in the manufacture of the bolt blank was the extrusion process. By this method, wire is used as large as, or larger than, the nominal diameter of the bolt and the portion to be threaded is extruded or reduced in diameter. By using this process longer bolts can be roll threaded and still have a full diameter shank than is possible by upsetting the shank. On low carbon steels which are not to be heat treated, cold working of the steel by the extrusion process increases the tensile and fatigue strength of the threads.

Continued improvement of roll threading machines, roll threading dies and bolt material has made it possible to roll almost any type of thread. For many years rolled threads have been used on some precision work as it has been found easier to maintain accurate thread dimensions by rolling than by cutting. Many taps are rolled and some micrometer screws.

One of the most recent developments in roll threading is the manufacture of aircraft bolts and studs with rolled threads. These threads before the war were manufactured by grinding from a solid bar. In this field alone the thread rolling process has made possible production of hundreds of millions of bolts and studs on which the accuracy of pitch diameter, form, lead and all physical properties are kept within rigorous aircraft standards. If these bolts and studs were to have been made by grinding, an investment in thread grinders and labor hours many times that needed for the rolled product would have been required.

Obviously, roll threading has not been as easy and simple as it may appear from the above discussion. When high carbon and alloy steel threads are to be made to close tolerances many of the defects to be found in steel products may add complications. Such pitfalls as decarburization, seams, and laps and "drunken threads" have had to be overcome by extreme care in manufacturing and close inspection. The experience gained in the manufacture of threaded fasteners for highly stressed parts in aircraft or automobiles has been used to enable bolt manufacturers to produce better fasteners for machinery for the farmer, road builder and miner. The roll thread process has come up a long way from the original inferior stove bolts and machine screws made back in the 1880's to the precision rolled thread aircraft engine studs now in use.

Cold Driving of Large Rivets

by W. E. Fowler, Jr.
Engineer, Riveting Apparatus, Inc.

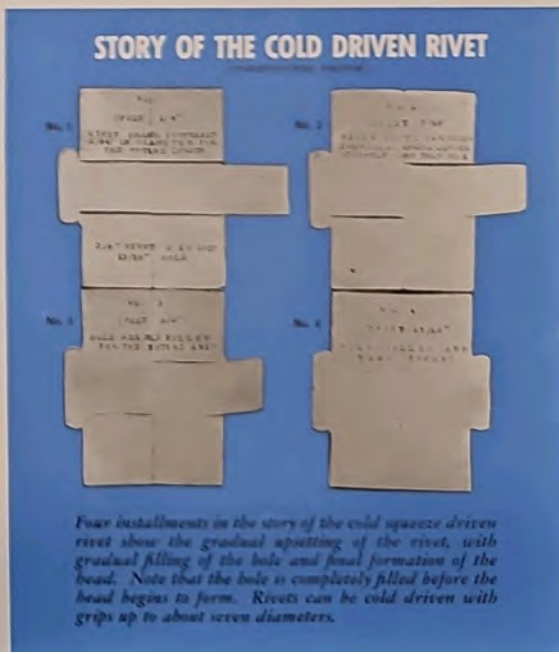
. . . . . cold riveting gives tight and rigid joints, producing a superior job at considerably lower cost . . . . the process is easily adaptable into many shops . . . .

LARGE cold driven rivets have been used to an increasing extent in many types of structures both in this country and Canada. More recently, large low-carbon alloy-steel rivets have been successfully driven cold in certain types of armament subjected to severe ballistic shock. In some cases these alloy rivets have been in the annealed condition and in other cases have been quenched and drawn. Although not yet widely practiced, cold driving of large rivets up to 2 in. in diameter has been found to be satisfactory and to produce sound and tight joints.

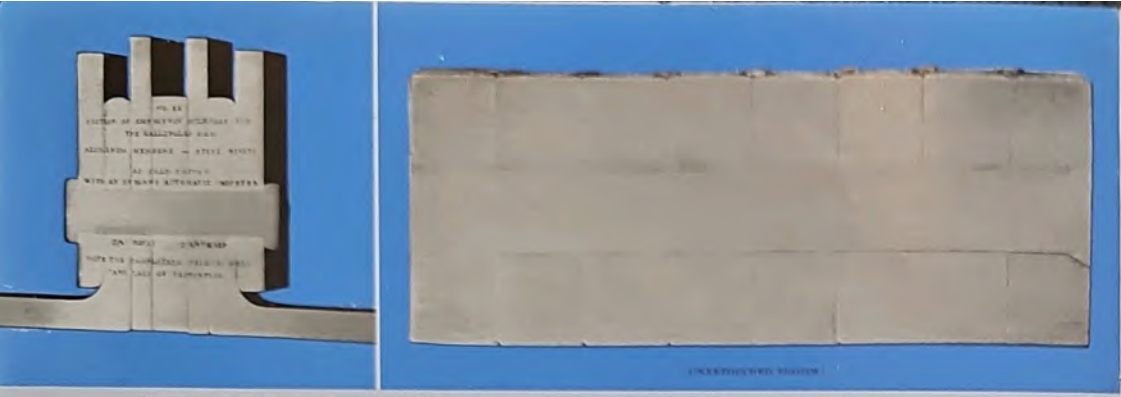
Cold rivets of ½ inch diameter and larger must be squeeze driven to obtain best results as the impact pressure of hammer driving merely peens over the head without forcing sufficient metal into the hole to fill it. Either pneumatic or hydraulic squeeze riveters may be used. Steady pressure applied to the cold rivet causes the shank to swell uniformly throughout its entire length, pressing outward against the surrounding metal so that it completely fills the hole and permanently accommodates itself to any irregularities in the hole surface or slight mismatching of holes. Resistance of the surrounding parts to further flow of rivet metal into the hole permits pressure to develop which simultaneously upsets the head and squeezes the plates together solidly. Release of the driving pressure after completion of the riveting cycle does not affect this intimate contact between the rivet and the surrounding metal so that a tight fit is obtained.

One of the greatest engineering values of cold driven rivets is the improved elastic characteristics of the joint. Because the rivets completely fill the holes, there is no play between rivets and the surrounding metal. The full resistance of the joint is available immediately and slippage or yielding cannot occur until the load reaches about 85 per cent of the ultimate shear strength, or about 38,000 psi for ordinary structural rivets. This is four times the usual design stress. The joint, therefore, is solid and rigid and withstands permanent distortion up to loads far in excess of the ordinary working stress in the structure.

Special manufactured rivet heads or rivets made from special steel are not required for cold riveting, as any convenient type of manufactured rivet head may be used, and standard rivet specifications, such as ASTM A141—Structural Rivet Steel, are suitable.



Four installments in the story of the cold squeeze driven rivet show the gradual upsetting of the rivet, with gradual filling of the hole and final formation of the head. Note that the hole is completely filled before the head begins to form. Rivets can be cold driven with grips up to about seven diameters.



Steel rivets, 3/4 in. in diameter, cold driven in aluminum and steel members, completely fill the holes and produce a rigid assembly. Caulking is unnecessary and the rivet is leak-proof. Note that the flow of metal produces a fillet under the driven head, adding to strength and resistance to fatigue.

Flat or modified cone heads are generally recommended for cold driving, as button or other conventional styles of driven heads require excessive pressure to upset. Most of the pressure in driving button and similar heads is lost by restraint of the die walls to free lateral flow of the metal. Considerably less pressure is required if the metal is unrestricted sideways and is permitted to flow freely by driving with flat smoothly polished dies. The resulting flat heads of diameter 1½ times and thickness ¼ times the shank diameter of the rivet are large enough to develop the full strength of the rivet.

The pressure required for cold driving a ¾ in. diameter rivet is about 45 tons for ordinary steel and somewhat higher for nickel or other high strength steels. Pressures for other size rivets are in proportion to the shank area. This pressure does not damage the rivet metal or riveted parts, but in order to avoid accidental excess pressure and to permit a quicker driving stroke it has been found advantageous to use some type of control such as the Osborne Automatic Pressure Control.

The clearance in rivet holes for cold riveting should be held as close as practicable and need not be as great as for hot riveting. There is no temperature expansion of the rivet to be allowed for and since the rivet is cold it can be handled and inserted easily and with less clearance. The small clearance is a distinct advantage since the rivet is guided better while driving and upsets with a truer and more concentric head.

Increased production rates and impressive material, labor and equipment savings as high as 25 per cent of the total riveting cost are obtainable by cold driving large rivets with controlled pressure. This is made possible because of the factors outlined below:

1. Use of controlled pressure reduces distortion and subsequent straightening operations.
2. Less bolting and fitting up since slight gaps between parts will be closed in riveting without washering the shank of the rivet or upsetting fins between members.
3. Fuel and heating equipment eliminated.
4. Less rivet stock in flat driven head.
5. Lower die cost due to simple design, less expensive die steel, and longer life of cold driving dies.
6. Eliminates loose or badly formed rivets.
7. Eliminates loss of overheated rivets.
8. Less compressed air is consumed.
9. Greater production by elimination of effects of furnace heat on workmen, ease of handling cold rivets, control of driving of rivets which are upset with one stroke and no holding pressure.

Cold riveting can be introduced easily into many shops with the savings indicated above. Unlike many processes the adoption of which requires large expenditures for equipment, personnel training and testing, cold riveting utilizes press riveters already installed in most shops and skill of men already trained in riveting. Cold riveting can be easily adopted and at considerably lower cost produces a superior job. Holes solidly filled give tight and rigid joints.

Tentative Specifications

For Cold Riveted Construction

Rivets ½ inch diameter and larger . . . September, 1942

The American Institute of Bolt, Nut and Rivet Manufacturers and the American Institute of Steel Construction in 1940 appointed a joint committee to investigate the use of cold-driven rivets and prepare a specification covering their use. The following specifications were developed and approved by the committee and are offered to the fabricating industry in tentative form for a period of trial. Suggestions for improvement of the specifications are invited. Copies may be obtained by writing the American Institute of Bolt, Nut and Rivet Manufacturers.

1. GENERAL

Cold riveted construction shall conform to the present specifications for riveted work and to the following requirements.

2. RIVETS

(*a*) *Grade.*

Rivets shall be of the same grade as used for hot riveting, such as ASTM A141 or A31, except that hot made rivets shall not be quenched, and cold made rivets shall be annealed.

(*b*) *Heads.*

Button, high-button, or any other American Standard type of manufactured head may be used. If approved by the customer, flat heads may be used having a diameter 1½ times and a minimum height of head ⅜ times the nominal diameter of the rivet.

(*c*) *Points.*

Rivet points shall be free from shearing cracks, and shall be reasonably square to prevent bending of the rivet shank during driving.

3. FABRICATION

(*a*) *Driven Head.*

The driven heads of rivets shall be flat head or a modified cone, or other types within the range of cold riveting as agreed upon between the user and the supplier. The diameter of the flat head shall be approximately 1½ times and the minimum height of head ¼ times the nominal diameter of the rivet; the modified cone head shall conform to the accompanying sketch.

(*b*) *Riveting Pressure.*

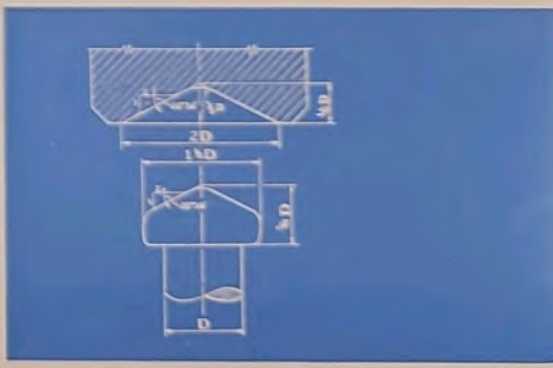
The pressure shall be properly controlled and sufficient to form the specified head, but not of an amount that will warp or buckle the work.

(*c*) *Driving Die.*

The die for driving the heads shall not restrict the free transverse flow of metal at the edge of the head.

(*d*) *Holes.*

Diameter of hole shall be larger than nominal diameter of rivet by not more than: 1/32" for ½" to ⅞" diameter, inclusive; 1/16" above ⅞".



Too Many Types?

To the casual observer, it might appear that the "standards" for ordinary hexagon and square bolts and nuts are excessive in number. To the experienced designer, however, each type of these products serves a definite useful purpose and is the result of a long process of evolution. In fact, the types as standardized by the trade are a direct result of the variety of user's requirements—requirements as to size, application, service, material, tolerance, appearance, etc.

A closer inspection reveals that the established standards follow an orderly, well defined pattern.

STANDARD BOLT HEADS

Wrench head bolts are divided into two series of heads, *regular and heavy*. The regular series, designed toward economical use of material with ample margin of safety, is recommended for general use. Heavy bolt heads, somewhat higher and wider than the regular type, are used only where greater bearing surface or wrenching surface is needed.

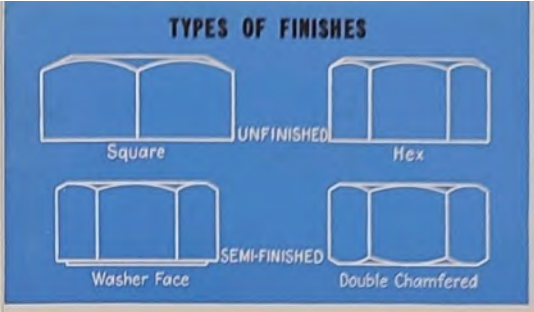
Two standard finishes are specified for each series of bolt heads. Unfinished heads, both square and hexagon, are left in the as-forged or as-upset (cold or hot) condition, and are not machined. Semi-finished heads are provided only in the hexagon shape and are similar to the unfinished heads, but have a bearing surface machined or otherwise treated to give a round $\frac{1}{64}$ in. thick boss under the head. This smooth bearing surface, practically square with the axis of the bolt, permits application with closer tolerances.

Cap screw heads, standardized in full finish, are washer-faced and have all surfaces of the body and hexagon machined or otherwise treated to provide an equivalent appearance. Provision has also been made for variations from this finish, subject to agreement between user and manufacturer.

Set screw heads (square) have also been standardized but do come under any other bolt head classification.

STANDARD NUTS

Four main series of nuts have been standardized; *regular, heavy, light* and *machine screw* (or *stove bolt*). As in the case of bolt heads, the regular series is intended for general purpose use, while the heavy series, slightly thicker and wider, offers increased bearing and wrenching surfaces. The light series, narrower than the regular, is used where space or weight must be held to a minimum. Four principal types of light nuts are provided, all of the same width but differing in height; light jam nuts, light nuts, light thick nuts and light castle nuts. Jam nuts are also specified for the other series, but only in the hexagon shapes.



Regular and heavy nuts, both square and hexagon, are obtainable in the unfinished condition. Regular, heavy and light series are provided in the semi-finished condition in hexagon form only. Semi-finished nuts may be either washer-faced, similar to the semi-finished bolt heads, or chamfered on the bearing surface, as shown in the accompanying sketch. In either case, the bearing surface is smooth and, to all practical purposes, square with the axis of the nut.

Slotted nuts, for locking cotter pins, are provided in the semi-finished hexagon form of all series. Slots are the same for all series, and may be either round or square at the bottom.

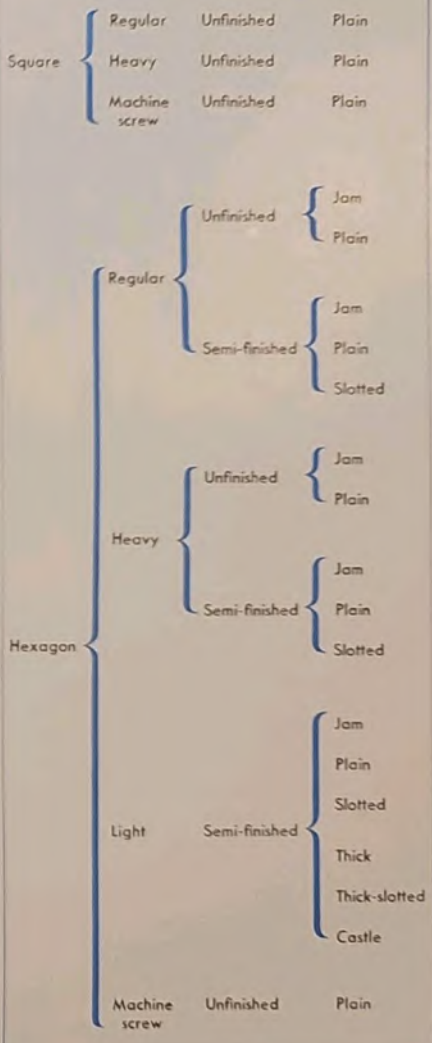
Castle nuts are slotted semi-finished hexagon nuts in the light series only. Instead of being chamfered on the top, the nut is cylindrical down to the bottom of the slot, where it is filleted out to the hexagon shape. Castle nuts are often used in place of the slotted nuts for appearance.

Machine screw and stove bolt nuts are small nuts in sizes No. 0 to $\frac{3}{8}$ in., either square or hexagon. Square nuts are flat on both top and bottom, while the hexagons are chamfered on top and flat on the bottom. For special purposes, and when so specified, the bearing surface may be chamfered or washer-faced.

★ ★ ☆ ★

The development of definite standards for bolt heads and nuts has materially reduced the variety of types as manufactured prior to the 20th century. It has led to a simplification of products universally made to standard dimensions and limits, although perhaps by widely divergent methods. It has made it possible to produce and to use these products with a minimum of confusion, supervision and wasted effort. Specifications should be made to these standards wherever possible.

Standard Square and Hexagon Bolts



If it is a standard

Industrial Fastener

you'll find
it in this Book

ARE THE BOLTS and nuts you use best fitted to your product and production methods? Do you know that rivets, bolts and nuts, although standardized to a point of becoming a shelf item of hardware, are nevertheless available in many various types to meet definite useful purposes? A bolt no longer is just an "iron rod with a knob on one end and a wrinkle on the other!"

To aid production in manufacturers' plants and to acquaint users of bolts, nuts and rivets with the accepted standards of types, diameters, lengths, materials and finishes, The American Institute of Bolt, Nut and Rivet Manufacturers has compiled a handbook of bolt, nut and rivet standards.

These standards have been variously developed over a long period of years by the American Standards Association, the Division of Simplified Practice of the United States Department of Commerce and the Institute, with the cooperation of manufacturers and users. In all cases, the standards are generally approved and represent established practices—practices that are practical and economical in manufacture, and that give interchangeable products satisfactory for the services intended. Tables are also given showing specifications for thread fits, dimensions, packing and weights for standard bolts, nuts and rivets.

This handbook can be obtained from the American Institute of Bolt, Nut and Rivet Manufacturers, 1550 Hanna Building, Cleveland 15, Ohio, at a cost of $1.00 per copy

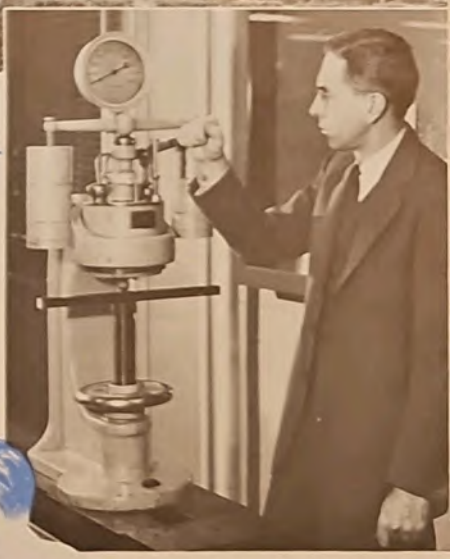
Scientific *Research*

helps solve problems for users of Fasteners

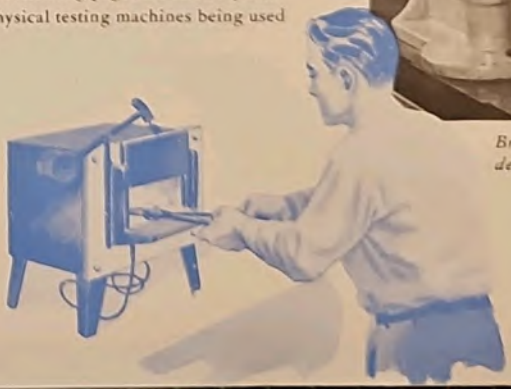
General view of The Technological Institute of Northwestern University.

RECOGNIZING the desirability of conducting technical studies that will supplement the work done by individual manufacturers and users of fasteners, the Institute has launched a comprehensive Research program.

The facilities of The Technological Institute of Northwestern University — one of the nation's most modern and best equipped laboratories — will be employed in this activity. On this and the following pages are shown photographs of some of the physical testing machines being used in these investigations.

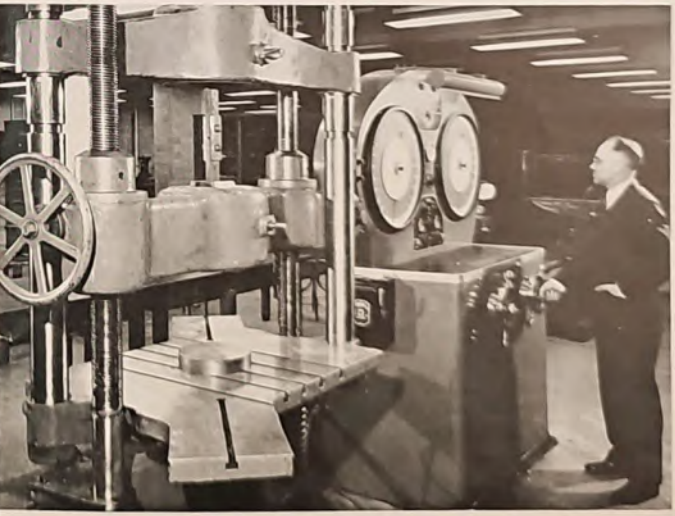


Brinell Testing Machine used to determine the hardness of specimens.



This two-and-a-half story testing machine, capable of exerting a pressure of 1,000,000 pounds on a beam 55 feet long, can measure loads to an accuracy of 1/10 of 1%. With this huge machine, full scale tests of large samples can be made.

Part of the machine shop where specimens are prepared for testing. This is typical of the fine modern equipment used throughout.



This 120,000 lb. machine tests specimens in tension, compression and flexural failure. For certain specimens, it can draw the stress-strain curves.

The Rieble Impact Machine measures stresses up to 200 ft. lbs. of energy in tension, shear and bending.

How Tight Should a Bolt Be?

An engineering discussion of the strength of highly stressed, dynamically loaded Bolts and Studs

by J. O. Almen

Research Laboratories Division, General Motors Corp.

A PAPER PRESENTED AT A RECENT MEETING OF THE SOCIETY OF AUTOMOTIVE ENGINEERS IN DETROIT

(Part I of a two-part article)

THE STRENGTH of most highly loaded bolts and studs is determined by the man with the wrench and not by the designer, the metallurgist, or by the manufacturing processes. In highly loaded bolts and studs we rarely find the design so bad, the fabrication practice so poor, or the material so weak as to cause failure in service providing that the nut is properly tightened against reasonably rigid members.

There are, of course, advantages to be gained by careful design. Good materials and heat treatments should be used; some manufacturing practices are better than others but all of these are relatively unimportant in severe service in comparison to proper tightening of the nut during assembly and to maintaining that tightness during service. We should, therefore, be less concerned with design details and give more attention to the technique of tightening nuts and to the ability of the bolted assembly to maintain the required tightness.

An estimate of the relative responsibility for the strength of bolts and studs, among the individuals who are concerned from design to assembly, is shown in Figure 1.

Fig. 1—Relative responsibility for fatigue durability of bolts.

A properly tightened nut is one that applies a tension load to the bolt or stud that is equal to or greater than the external load that is to be supported in service. When this condition is fulfilled and maintained against reasonably rigid bolted assemblies, the bolt can not fail by fatigue because it can experience practically no change in stress regardless of the fluctuating nature of the operating load. It can not fail statically because, to be tightened as specified, it must be capable of supporting the greatest operating load.

FATIGUE STRENGTH VARIES

Numerous fatigue tests have shown that the fatigue strength of metal decreases as the range of dynamic stress (stress change) to which the metal is subjected is increased and, conversely, that the fatigue strength is increased as the dynamic stress range is decreased. As the stress change approaches zero, the dynamic load that can be supported approaches the tensile strength of the material.

Consider the case of a bolt subjected to fluctuating load, such as in a connecting rod. If the nut is tightened just enough to make contact with the bearing cap, the load that will be applied to the bolt will vary from zero, the initial tightness, to the maximum inertia load of the piston and the connecting rod. Under this large stress change, the strength of the bolt will be less than one-fifth of its strength under static load and fatigue failure can only be avoided by making the bolt very large. If we now tighten the nut so as to load the bolt to one-half of the inertia tension load, the stress change will be one-half as great and its operating strength may be one-third to one-half of its static strength. When the nut is tightened against reasonably rigid abutments to produce a tension in the bolt, equal to or

greater than the working tension load, there is practically no stress change in the bolt and its operating strength, therefore, approaches its static strength.

BOLT FATIGUE TESTS

The increased fatigue strength of bolts as the initial tightness is increased is shown by a series of fatigue tests on bolts that were dimensioned and processed as shown in Figure 2, in which the external load applied to the bolted members alternated from zero to 9215 pounds. One group of 17 bolts was fatigue tested in which the nuts were tightened to produce initial bolt tension of 1420 pounds. The load in these bolts, therefore, alternated from a minimum load of 1420 pounds to a maximum load of something more than 9215 pounds, a load change of nearly 8000 pounds at each load application. The average fatigue durability of these bolts was 5960 cycles.

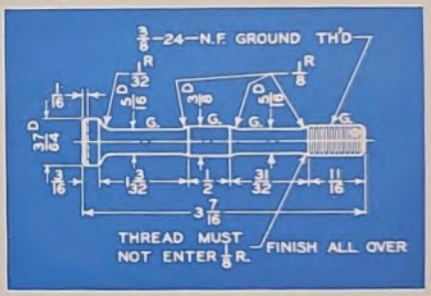


Fig. 2—Test bolt detail.

Another group of 16 bolts was fatigue tested in which the nut was tightened to an initial bolt tension of 5920 pounds. The load in this group, therefore, ranged from 5920 pounds to 9215 pounds or a load change of approximately 3300 pounds at each load application. The average fatigue life of this group of bolts was 35,900 stress cycles.

In a third group of 15 bolts the load alternated between the initial nut tension of 7220 pounds and 9215 pounds, a load change at each load application of about 2000 pounds. The fatigue life average of this group was 214,500 stress cycles.

In the fourth group only two bolts were tested, one failing after 4,654,000 stress cycles and the other had not failed when the test was stopped after more than 10,000,000 stress cycles. These bolts were initially tightened to 8420 pounds and since the external load was the same as for the other groups, the load change at each load application was only 800 pounds.

The results of these tests are graphically shown in the chart in Figure 3, in which the initial bolt tension in pounds is plotted against the fatigue durability in stress cycles. The test points for the two bolts tightened to 8420 pounds initial tension are not shown because their inclusion would require contraction of the life scale to undesirable proportions.

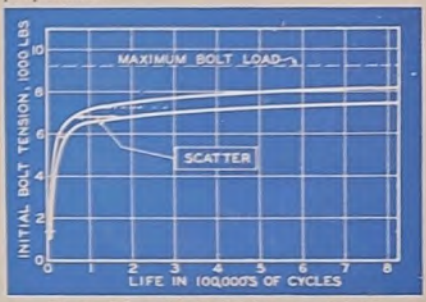


Fig. 3—Effect of initial tightness on bolt life.

These test data are also shown in the log-log plot in Figure 4, in which the load or stress range to which the bolts were subjected is plotted against durability in stress cycles. For this plot the stress range R is circulated as:

$$R = 1 - \frac{\text{Minimum Stress}}{\text{Maximum Stress}}$$

Log-log plotting is used because the lines joining the test points are only slightly curved which permits estimating the trend of the curve between test points and also permits projecting the curve into the region of the fatigue endurance limit.

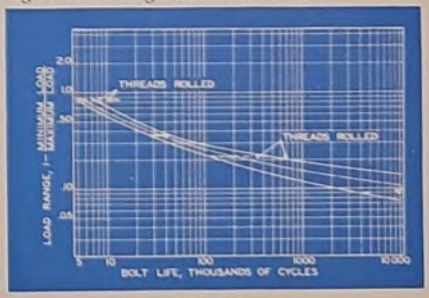


Fig. 4—Bolt fatigue curve.

Note that there is a considerable variation in the durability of the bolts in each group of bolts tested. Similar fatigue life variability occurs in all kinds of specimens because it is impossible that any specimen can be exactly like any other specimen, just as no person can be exactly like any other person. The fatigue life variability among these bolts, as is shown in Figure 4, is small as compared to the life variability of other types of machine parts. The ratios of life of the best to the poorest in each group, in decreasing order of stress range, are 1.5, 1.65 and 4.0. Although there are not a sufficient number of tests in each group to definitely establish the limits of the scatter band, increased scatter with increased life is to be expected from the nature of the fatigue curve. The data on bolt fatigue tests that have been given above are summarized in Table 1.

STRESS FLUCTUATION UNAVOIDABLE

When the nut is insufficiently tightened, each load application will elastically elongate the bolt and a gap will be formed between the bolted members, the magnitude of which will be reduced as the initial tightness is increased. When the nut is tightened to near the operating load, this separation of the bolted members will not occur, but there will, nevertheless, be a small elastic elongation of the bolt.

When the nut is tightened, the bolt is elastically elongated and, at the same time, the bolted abutments are elastically shortened; the amount of shortening depending upon such variables as the design, the elastic characteristics of the bolted materials, and the area of the material supporting the bolt load. On application of the external operating load, the bolt is acted upon by the external load plus the load of elastic recovery of the bolted assembly. The sum of these forces will always be greater than the initial bolt tension and, therefore, the bolt will experience a change of stress at each load application regardless of the magnitude of the initial tension. The stress change will be greater as the elastic deformation of the bolted assembly is increased and it will be reduced as the rigidity of the bolted assembly is increased.

STRESS–STRAIN MEASUREMENTS

As a preliminary to the fatigue testing of the bolts described above, stress-strain measurements were made for each bolt. The fatigue machine used for these tests is equipped with gravity scales whereby the load applied to the specimen may be accurately weighed at any position of the load applying device. It was, therefore, a simple matter to take stress-strain measurements on each bolt while in place in the machine by weighing the load and measuring the elongation of the bolt at frequent intervals.

A typical plot of stress-strain data taken from one of the test bolts is shown in Figure 5. In this plot the inclined straight line, having its origin at zero load and zero elongation, is the load extension curve of the bolt alone (curve A). The plotted points, curve B, were taken from a bolt that had been tightened against relatively rigid abutments to an initial load of 4000 pounds, under which load the bolt elongated 0.0046 inches. As the external load was applied to this tightened bolt, the load extension curve B, as shown by the plotted points, does not rise vertically to join the inclined load extension curve of the bolt alone (A) as would occur under ideal conditions of an elastic bolt and perfectly rigid abutments. The plotted points (B) join the vertical line at 0.0046 inches elongation and the inclined line (A) by an easy curve thus showing that the bolted members are elastic and that the load on the bolt is augmented by the elastic recovery of the bolted members.

TABLE 1
BOLT FATIGUE TEST DATA

Initial Tension	*Operating Load*	*Load Change*	*Stress Range*	*Average Life*	*Life Variation*
1420	0 to 9215	8000	.85	5,960	1.5
5920	0 to 9215	3300	.36	35,900	1.65
7220	0 to 9215	2000	.215	214,500	4.
8420	0 to 9215	800	.087	5,000,000+	———

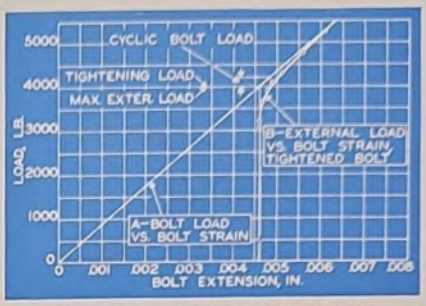


Fig. 5—Determination of cyclic bolt load.

EFFECT OF ELASTICITY

This added load increment due to the elasticity of the bolted members can be measured from the curves. For the case in which the external load is equal to the initial bolt tension, extend the bolt pre-load (4,000 pounds) to intersect the curve of the plotted points B which will be found to be at a bolt extension of 0.00485 inches. Since the extension under the bolt pre-load was 0.0046 inches the bolt elongation has now increased 0.00025 inches. Now project the new bolt extension line (0.00485 inches) vertically to intersect the load extension curve (A) of the bolt alone. This will correspond to a load of 4220 pounds or a load increase of 220 pounds over the initial bolt tension.

The load range experienced by the bolt under these conditions would be $1 - \frac{4000}{4220} = 0.052$, a stress range so small as to be practically zero (see Figure 4) and the bolt load is, therefore, almost static. In like manner, the resultant bolt load may be found for any external load whether it be greater or less than the initial bolt tension.

The plot Figure 5 was made from a fatigue test fixture in which the bolted members were relatively massive and rigid and care was used to assure parallelism and smoothness of the contacting surfaces. For these reasons the elastic compression curve of the bolted members is so steep as to be immeasurable and it, therefore, coincides with the vertical pre-load line at 0.0046 inches bolt extension up to a load of about 3400 pounds. From this point the elastic curve (B) swings to the right and joins the bolt elastic curve (A) at about 0.0060 inches bolt extension and 5220 pounds load. The elastic compression curve for this bolted assembly is such that external loads up to 3400 pounds add very little to the bolt pre-load. The non-linear portion of the curve is presumably due to bending of the bolted members resulting from imperfect surfaces and to bending of the loading beam under the external load. As the external load is increased through the non-linear range, the cyclic increment of bolt load above the initial bolt tension increases rapidly until complete separation of the bolted parts occurs at 5220 pounds and 0.0060 inches bolt extension.

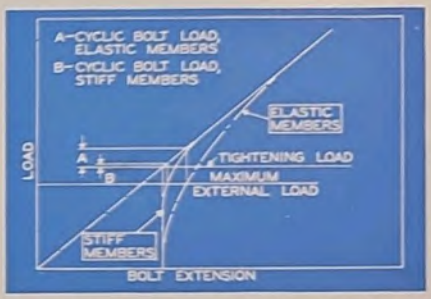


Fig. 6—Effect of elasticity of bolted members.

The schematic chart Figure 6 shows two elastic curves to illustrate the effect of varying the elasticity of bolted members. From the method of analysis described above, it will be seen that the cyclic load on the bolt is increased as the elasticity of the bolted members is increased. Similarly, Figure 7 shows that the cyclic bolt load decreases as the elasticity of the bolt is increased.

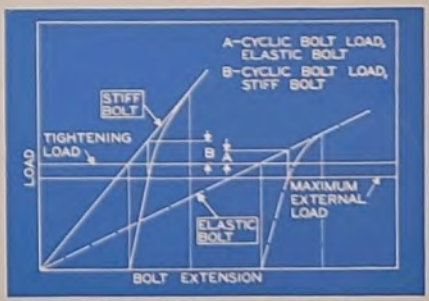


Fig. 7—Effect of elasticity of bolt.

The strength of a bolt may, therefore, be said to be influenced by the design of the bolted assembly and the materials used in the assembly. A properly tightened bolt will be weaker when used in combination with aluminum abutments than when used in combination with steel abutments of similar proportions because of the greater elastic deformation of aluminum. It will be weaker when used against abutments of small area than when used against abutments of large area. The number of parts making up the bolted assembly will also affect the bolt strength because mating parts will always fit imperfectly, thus adding bending deflection to elastic compression, and each part will, therefore, add to the total elastic deformation of the assembly.

ARRANGEMENT AFFECTS BOLT STRENGTH

The arrangement of parts in a bolted assembly can have great effect on bolt strength. An exaggerated case of this kind is shown in Figure 8, in which identical parts are used in two assemblies, in one of which the bolt would be weak and in the other, the bolt would be strong.

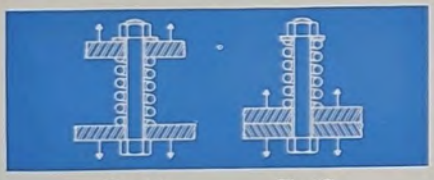


Fig. 8A *Fig. 8B*

In Figure 8A, a bolt is tightened against two stiff plates that are spaced apart by a low rate spring. The external load is applied against the bolt by the two stiff plates. In this case the coil spring is a part of the bolted assembly and since it is very elastic as compared to the elasticity of the bolt we may, for purposes of discussion, consider its rate to be zero. When an external load equal to the initial bolt tension is applied to the plates, the total bolt load will be twice the initial tension, because the spring load will not be appreciably reduced by the additional extension of the bolt and its load will be added to the external load. The stress range experienced by this bolt will, therefore, be $1 - \frac{1}{2} = 0.5$. For the stress magnitudes of the bolts shown in Figure 4, it will be seen that at stress range 0.5 early failure may be expected.

In Figure 8B, the external load is also applied to the plates, but the low rate spring now is a part of the bolt instead of a part of the bolted assembly as in Figure 8A. We, therefore, have a very elastic bolt tightened against relatively very rigid members. When an external load equal to the initial bolt tension is applied to the plates, the load between the plates will be reduced to zero but the bolt will experience no appreciable change in load. The stress range will be $1 - 1 = 0$; that is, the load will be static and fatigue failure can not occur (see Figure 4).

Fig. 9—Bolts subject to fatigue.

PRACTICAL ELASTIC CONSIDERATIONS

While the arrangements shown in Figure 8 are rather fanciful, we have assemblies in actual practice that are somewhat similar. Figure 9 shows a condition frequently met in practice in which a gasket is clamped between two mating plates. The gasket is compressed locally opposite the bolt, thus bending the plates until they serve as the spring shown in Figure 8A. When the external load is applied, the bolt will feel the elastic load of the bent plates plus the external load, a condition that invites trouble.

Also in such assemblies the bolt or stud is usually short and relatively inelastic and we, therefore, have all the conditions necessary for large stress range and resulting fatigue failure. When gaskets are necessary, the gasket area should be large in the vicinity of the bolt and relatively smaller in proportion to the distance from the bolt, in order that the unit pressure on the gasket may be maintained with minimum bending of the bolted surfaces. The gasket should be as inelastic as possible and still serve its purpose.

Fig. 10—Spring augments bolt elasticity. Bolt life is increased.

A practical assembly having the characteristics of the assembly in Figure 8B is shown in Figure 10, in which a spring washer is clamped between the bolt head or nut and the bolted assembly. When such a spring washer is dimensioned to support the required external load or the initial bolt tension, whichever is the greater, within the elastic range of the washer, the effective bolt elasticity is increased and, therefore, the bolt fatigue strength is increased.

Likewise any other expedient that will increase the rigidity of the bolted assembly or that will increase the elasticity of the bolt will increase the fatigue strength of the bolt.

TIGHTNESS MUST BE MAINTAINED

Although properly tightened bolts are stronger as the rigidity of the bolted assembly is increased and as the elasticity of the bolt or stud is increased, a bolt or stud that is incapable of maintaining an initial tension equal to the external tension load is quite likely to fail in severe service. Thus we find relatively greater mortality among short bolts and studs than among long ones because of their lesser elastic yield.

Consider a short stud such as is often used to attach engine cylinders to the crankcase. The thickness of the cylinder flange plus the thickness of the washer may be less than one-half inch. When the nut is tightened, the stud is elastically elongated through the length from near the bottom of the nut to near the last thread engaging the crankcase. Assuming that this distance is one half-inch and that the nut, at proper tightness, stresses the stud to 120,000 psi, the stud will then be elastically elongated 0.002 inch. If during operation, the bolted assembly is reduced in the thickness by 0.001 inch through wear, corrosion, embedding, or by displacement of material such as soft plating, the stud will lose one-half of the required tension and fatigue failure will follow.

Greater safety is assured when longer studs are used. For example, when under the same conditions as described above, a stud of two inch effective length is used, the loss of tension will be only one-eighth of the initial tension; that is, a loss of 0.001 inch from a total elastic elongation of 0.008 inch. This would be serious only when other factors, such as initial tension, are near the low limit.

It will, therefore, be seen that among the major bolt and stud hazards, particularly when they are short, must be included loss of thickness of the bolted assembly as a result of surfaces that can embed, soft plated coatings that can be displaced, gaskets that can be plastically compressed, materials that can yield at elevated operating temperatures, etc.

YIELD POINT MAY BE EXCEEDED

In tightening nuts it is far better to stress the bolt above the yield point of the material than to risk undertightness. Since for adequate size bolts, the load is practically static, there can be no harm in yield providing that the plastic deformation is not so great as to reduce the static strength of the bolt. The amount of yield that can be tolerated will depend on the length of the bolt, the design of the bolt and the characteristics of the material. When the body of the bolt or stud is equal in diameter to the outside diameter of the threads, yield will be concentrated at the roots of the threads whether the bolt be long or short. In such cases little yield, as measured by bolt elongation, can be tolerated. For bolts or studs in which the body diameter is equal to or less than the thread root diameter, greater yield, as measured by bolt extension, can be tolerated since yield will occur over the entire body length.

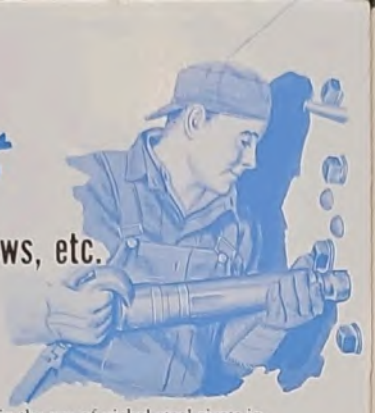
It is not intended to recommend that all bolts should be tightened above the yield point but only to show that, where necessary, the practice may be followed provided proper caution is used. The necessity for tightening above the yield point would occur in such cases as when the bolt yield strength is only slightly greater than the maximum external tension load or when cotters are used in combination with castle nuts. In the latter case, it is often necessary to advance the nut beyond the required tightness, in order to admit the cotter since the nut should not be slackened to meet the cotter pin hole. For short bolts, two cotter holes should be provided at such angles as to register with the notches in the nut at thirty degree intervals, thus reducing the overtightness in critical cases.

In the concluding part of this article, Mr. Almen discusses Bending Loads, Locking Devices, Nut Tightening Practices and other factors.

Fasteners Digest

Current Information on Bolts, Nuts, Rivets, Screws, etc.

as reported by leading Industrial Publications



Because of the many special problems allied with armament and products for war, new assembly practices and new fastening components have been developed in great variety. *Panel fasteners* are widely used in aircraft construction and depend for their locking power either on screw action alone or on the action of flat or helical springs. *Threaded inserts* are provided in various designs for use in wood, plastic or soft materials. *Rivets* have received considerable attention. Conventional types still find application in wartime production, but those which can be headed cold without bucking are widely applied. *Screws* with recessed heads are being adopted to a great extent. *Washers* find an augmented field of specification in war products where it is of greatest importance that a screw or nut stay where it is put. *Safety nuts* have been developed in numerous types, so constructed as to lock themselves in tightened position. It is believed that many of these newer fastening devices will find applications in future product designs.

"Wartime Fastening Devices Revise Assembly Methods", Electrical Manufacturing, November, 1943.

Most of the conventional joining methods, with minor adoptions, can be used for joining magnesium. Rivets of aluminum alloys that have little or no tendency to cause galvanic corrosion of magnesium are preferred to magnesium alloy rivets. Steel or brass rivets should be avoided. Cone point head rivets are less likely to damage magnesium sheets because of their relatively low driving pressure. With screw fastenings care must be taken in design to avoid stress concentration at bolt or screw holes. In general, a thread length equal to 2 or 3 times the diameter of bolt or screw should be provided. A coarse thread with flat or rounded top is preferred. Steel screws and studs may be galvanized or cadmium plated to assist in preventing galvanic corrosion.

"Fabricating Methods for Magnesium Alloys" — Product Engineering, September, 1943.

Experience in the use of nickel steel rivets in boilers has shown a tendency for such rivets to fail both in the shank and under the head. Some of these failures are attributable to conditions of heating and driving the nickel steel rivets, which seem to require extreme care in heating and cooling. Failure of these rivets reduced their holding power so that excessive movement of the sheets in the areas affected occurred, thus contributing to the failures of the plates which followed. Experience with an identical boiler in which carbon steel rivets were used has failed to disclose any similar difficulties. It is also observed that an increase in the hardness of rivets requires higher driving pressures, and these pressures in turn produce increased wall pressure on the rivet holes, thus tending to aggravate any tendency toward intercrystalline corrosion introduced by water conditions.

"Alloy Steel in Boilers", by A. G. Trumbull, Chief Mechanical Engineer, Chesapeake and Ohio Railroad, Cleveland, Ohio. Railway Mechanical Engineer, February, 1943.

Some aircraft designers avoid the use of threaded joints for primary aircraft structures because of possible stress concentrations arising from the notch effect at the bottom of the thread. Modification of the present American (National) form of thread to include a radius at the root paves the way for general use of threaded joints in such structures, and yet conforms to standardization requisites. It is suggested that the root radius be made equal to the maximum "worn tool arc" limits allowed by American thread standards. This, it is thought, is preferable to the use of the round root Whitworth thread. Many clever designs in aircraft structures can be made through the application of threaded connections.

"Modified Standard Thread for Primary Aircraft Structures", by Mario Di Giovanni, Project Engineer, Development Division, Curtiss-Wright Corporation. Product Engineering, October, 1943.

The magnitude of the production job involved in the manufacture of bolts, nuts, screws and rivets may be judged from the fact that more than 170,000,000 pieces are turned out each day, consuming about 2 per cent of all the steel produced. Included in this volume are vast numbers of standard items, but there are also great numbers of special items for various industries. In spite of tremendous peace time work on standardization within the industry itself — so effective that similar products of more than 100 plants are interchangeable — there are about 450,000 different items of bolts, nuts, screws and rivets manufactured and stocked in normal times. At present, under government order, a simplified stock list of 250,000 standard sizes prevails.

"Fastening Devices", by E. J. Tangerman, Technical Editor, Mill Supplies, March, 1943.

Indicative of the ingenuity applied to the wartime production in the fastener industry is the story of how National Screw and Manufacturing Company, Cleveland, Ohio, tackled a special nut problem. The unit was a dome nut for self-sealing fuel tanks and bomb bay doors on combat planes. The nut had usually been made from bar stock on automatic screw machines. The discovery of short cuts, however, relieved bottle-necks in production, saved machine-hours and man-hours, and saved 45 per cent in material. The new method, which entailed cold upsetting stock of smaller diameter, met the very close tolerance requirements of the product.

Machining stainless steel instrument screws from bar stock, with cut threads, as previously practiced, required the use of automatic screw machines (a critical bottle-neck), as well as a waste of critical material. National developed a method of manufacture from wire of smaller diameter, upsetting the head, machining the blank on non-critical equipment, and rolling the thread. This effected a saving of 61 per cent of the stainless and released machines badly needed for other purposes. Furthermore, the operations could be performed by unskilled workers with a minimum of training.

"Fasteners by the Million", by C. F. Neupher, vice president, National Screw and Manufacturing Company, Cleveland, Ohio. Steel Horizons, Vol. 5, No. 5, 1943.

The use of smaller diameter bolts, of heat treated stock so as to give the same strength, has been advocated as a means of increasing fastening efficiency, reducing weight and cost, and saving material. The torque required to load a bolt at a given stress per sq in. varies as the cube of the diameter of the bolt, and that necessary to produce a given grip or pull varies directly as the diameter of the bolt. Thus, standard wrenches, with a reasonable pull, produce relatively high strain in bolts of small diameter and a very low one in bolts of large diameter, or, conversely, smaller bolt diameters permit more effective fastening, since it is easier for an operator to pull up a smaller bolt to its maximum effective tension than to do so with a large bolt. Of course, for equal strength, the small bolt must have a higher unit tensile strength, which may be obtained by heat treatment.

"Heat-Treated Carbon Steel Bolts", by A. E. R. Peterka, Executive Engineer, Lamson and Sessions Company, Cleveland, Ohio. Steel, May 10, 1943.

The difficulty of handling the millions of small rivets used in airplane production has led to the development of an unusual magazine injector device, manufactured by the Cleveland Pneumatic Tool Company, Cleveland, Ohio. Christened the Cleco Riv-N-Jector, the device releases the rivets, one at a time, from the magazine when the operator presses it over the hole. The device is easily loaded with rivets, shank down. An injector spring exerts constant pressure on the rivet load to keep a rivet always in the jaws of the ejector. These jaws are spread by contact with the sheet being riveted, sufficiently to release the rivet. When empty, the device is reloaded at a "rivet center". Use of injectors of this type will prevent rivet waste. It is stated that only one out of three rivets issued is actually driven in airplane construction, and that 50 per cent of those dropped on the floor are not recovered in usable condition.

"Rivet Waste" — Steel, January 10, 1944.

Other developments brought about by the tremendous riveting job in the airplane industry are evidenced by a number of rivet sorting machines. For the greater part, a primary selector sorts the rivets according to shank diameters or thickness. The secondary assorting operation separates the rivets according to length and head type. A typical sorting machine, as variously developed by engineers of aircraft manufacturers, has a capacity of about 100 lb. of rivets per 8 hr. turn.

"Machine Sorts 200 Pounds of Rivets Per Day" — Steel, March 29, 1943.

"Automatic Sorters for Aircraft Rivets", by H. O. Mills and Ludwig Donner, Consolidated Vultee Aircraft Corporation, American Machinist, July 8, 1943.

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